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NATURAL CONVECTION HEAT TRANSFER BETWEEN ARRAYS OF HORIZONTAL CYLINDERS AND THEIR ENCLOSURE

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ABSTRACT

The natural convection heat transfer between arrays of horizontal, heated cylinders and their isothermal, cooled enclosure was experimentally investigated. Four different cylinder arrays were used: two in-line and two staggered. Four fluids (air, water, 20 cs silicone, and 96% glycerine) were used with Prandtl numbers ranging from 0.705 to 13090.0. There was no significant change in the Nusselt number between isothermal and constant heat flux conditions of the cylinder arrays. The average heat transfer coefficient was most affected by the spacing between cylinders and the total surface area of the cylinder arrays. The enclosure reduced the expected increase in both the average and the local heat transfer coefficients caused by changing the inner body from an in-line arrangement to a staggered arrangement of comparable spacing. An increase in fluid viscosity reduced the influence of the geometric effects.

NOMENCLATURE

A_i	Surface area of the inner body
A_o	Surface area of the outer body
B	Length of boundary layer on one cylinder, $B = \pi(d/2)$
c_p	Specific heat at constant pressure
d	Diameter of a cylinder
g	Acceleration of gravity, 9.81 m/sec^2
h	Heat transfer coefficient, $h = Q_{CONV}/A_i\Delta T$
k	Thermal conductivity
L	Hypothetical gap width, $R_o - R_i$
Nu_X	Nusselt number, hX/k, where X is any characteristic length
Pr	Prandtl number, $c_p\mu/k$
Q_{CONV}	Heat transfer by convection
Ra_X	Rayleigh number, $\beta\rho^2 g(T_i - T_o)X^3 c_p/\mu k$, where X is any characteristic length
Ra_X^*	Modified Rayleigh number, $Ra_X^* = Ra_X(L/R_i)$
R_i	Radius of a hypothetical sphere equal in volume to the volume of one cylinder times the number of cylinders in the cylinder array
R_o	Radius of a hypothetical sphere equal in volume to the outer body
S	Characteristic length, $S = (R_o - R_i)(A_i/A_o)$
T_i	Inner body temperature
T_o	Outer body temperature
ΔT	Temperature difference, $\Delta T = T_i - T_o$
X	Any characteristic length
β	Thermal expansion coefficient
μ	Dynamic Viscosity
π	Ratio of circle circumference to its diameter (3.14159)
ρ	Density of the fluid

INTRODUCTION

Natural convection heat transfer from a body to an infinite fluid medium has received extensive experimental and analytical study in the past, while relatively little attention has been devoted to natural convection within an enclosure. The increase in complexity, caused by the strong interaction between the boundary layer and the adjacent fluid, has made it difficult to obtain a solution to the problem of natural convection in an enclosure. However, interest in this area is dramatically increasing because of the important applications it has in areas such as nuclear reactor technology, electronic instrumentation packaging, aircraft cabin design, crude oil storage tank design, solar collector design, and energy storage systems.

One of the first in-depth experimental studies of natural convection in enclosures was performed by Warrington [1]. The heat transfer from inner bodies such as spheres, cubes, and cylinders to both spherical and cubical enclosures was investigated.

Larson, Gartling, and Schimmel [2] used laser interferometry to experimentally determine the temperature field around a heated, horizontal cylinder in an isothermal, rectangular enclosure.

Dutton and Welty [3] conducted an experimental study of natural convection heat transfer in an array of uniformly heated vertical cylinders surrounded by a vertical, cylindrical enclosure with mercury as the fluid medium. Their results indicated that the natural convection heat transfer was strongly dependent on the cylinder spacing.

Van De Sande and Hamer [4] studied the steady and transient natural convection heat transfer between horizontal, concentric cylinders with constant heat flux surface conditions. Their experiment showed that a sidewise displacement of the inner cylinder did not affect the heat transfer results. However, the overall heat transfer decreased or increased depending on whether the inner cylinder was above or below the centerline of the outer cylinder.

Crupper [5] performed an experimental study of natural convection heat transfer between a set of four isothermal, heated cylinders and an isothermal, cooled, cubical enclosure to determine the effect of the positioning of the cylinders within the enclosure.

Powe [6] investigated the limits of relative gap width for which available correlation equations for natural convection heat transfer in enclosures were applicable. Heat transfer rates for large relative gap widths were shown to be limited by those obtained for free convection to an infinite fluid medium, and this criteria was used to calculate a maximum relative gap width for which the enclosure equations were applicable. A minimum realtive gap width for applicability of the enclosure equations was determined by the pure conduction limit.

Brown [7] experimentally studied the effects of reduced pressure on the natural convection from a cylinder and a cube to a cubical enclosure.

Powe, Warrington, and Scanlan [8] performed a detailed study of natural convection flow phenomena that occur between a body of relatively arbitrary shape and its spherical enclosure. Resulting trends in the fluid flow data were established to facilitate better predictions of the heat transfer in problems of natural convection in enclosures.

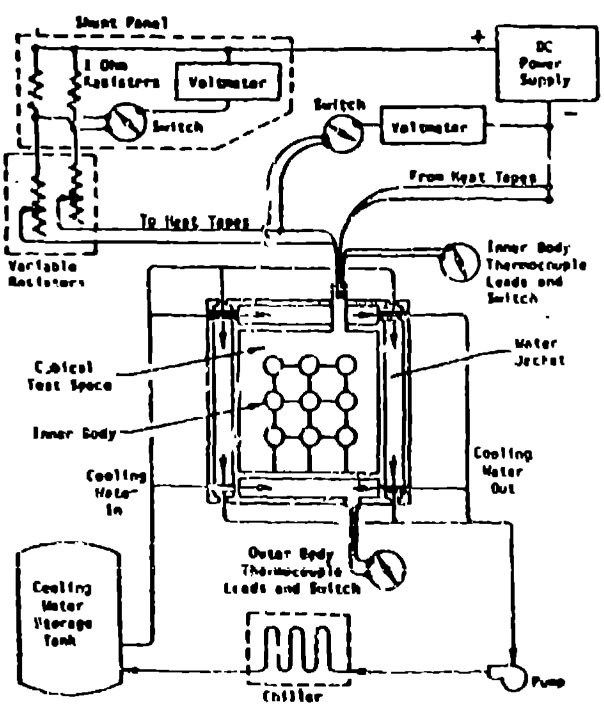

Figure 1. Schematic of the heat transfer apparatus

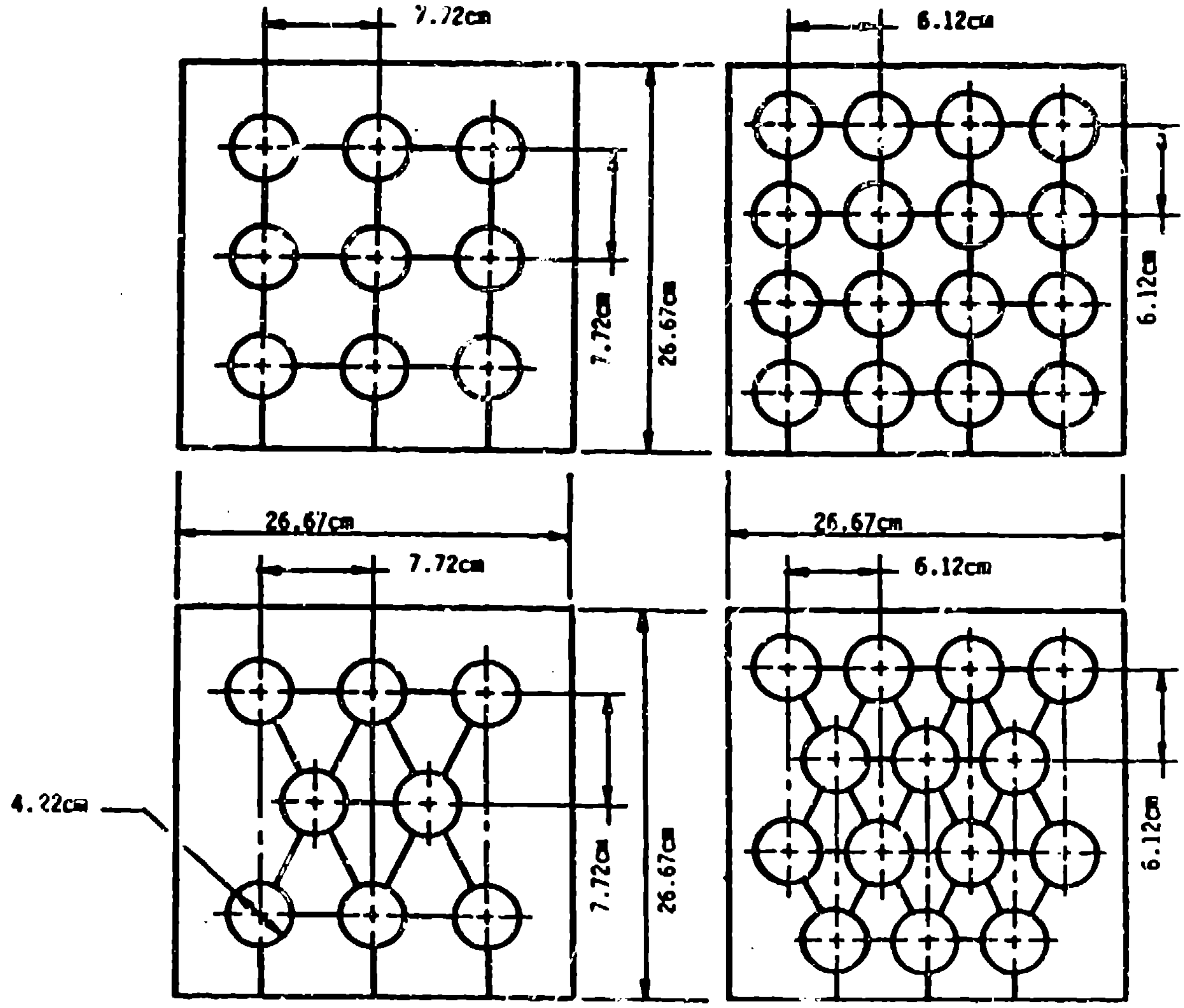

Figure 2. The four cylinder arrangements

The purpose of this study is to experimentally investigate the dissipation of heat by natural convection from arrays of heated, horizontal cylinders to a cooled, isothermal, cubical enclosure. The cylinders are subjected to both isothermal and constant heat flux conditions. Four fluids and four cylinder configurations are utilized. The fluids used are air, water, 96% glycerine, and 20 cs silicone. The four cylinder configurations consist of two in-line arrangements using nine and sixteen cylinders and two staggered arrangements using eight and fourteen cylinders.

APPARATUS AND PROCEDURE

The enclosure used for this investigation was a cube 20.07 cm along an inner side, constructed from 1.27-cm-thick, type 6061, aluminum. A water jacket enclosure, which measured 38.1 cm on a side, surrounded the cubical test space. The water jacket consisted of six separate rectangular channels each 3.175 cm in width, which gave one channel for each face of the test space. The flow of cooling water to each of the channels was fed by a manifold system and adjusted to maintain the cube that enclosed the test space at isothermal conditions. The cooling water was collected from the water jacket and pumped through a chiller into a storage tank and then back into the water jacket. A schematic of the apparatus is shown in Figure 1.

The four different arrangements of horizontal heated cylinders that were used for inner geometries are shown in Figure 2. The cylinders were constructed from 0.30-cm-thick copper pipe

19.46 cm long, and 4.22 cm outside diameter. Copper end caps 0.25 cm thick and 4.22 cm in diameter were mounted to both ends of each cylinder. The support structure for each array of cylinders consisted of wooden dowels 0.32 cm in diameter.

Heat was supplied to the inside surface of each cylinder with electrical resistance heat tape and a direct current power source. Input voltages to the heat tapes were controlled individually with variable power resistors that were connected in series with the heat tapes. The inside surface temperatures of the enclosing cube and the outside surface temperatures of the cylinders were monitored with embedded copper-constantan thermocouples.

Sixteen fluid/geometry combinations were used to obtain 172 data points, consisting of both isothermal and constant heat flux conditions for the inner geometries. After centering one of the inner geometries in the enclosure and filing the enclosure with one of the fluids, power was applied to the cylinders and cooling water was pumped through the water jacket. When equilibrium was established (approximately 2-4 hours) and the power was input to each cylinder, then the cylinder temperatures and the enclosure temperatures were recorded.

The heat transferred by natural convection was obtained by subtracting the heat transferred by radiation and conduction from the total amount of heat transferred. The conduction loss through the support structure, through the thermocouple lead wires, and through the heat tape lead wires was obtained with a one-dimensional analysis. Because the water, 20 cs silicone, and 96% glycerine were opaque to radiation, only the data using air as the fluid needed a correction for radiation. The radiation loss was experimentally obtained by evacuating the test space between the cylinders and the enclosure to a pressure below 50 μm and subtracting the conduction losses from the total power input.

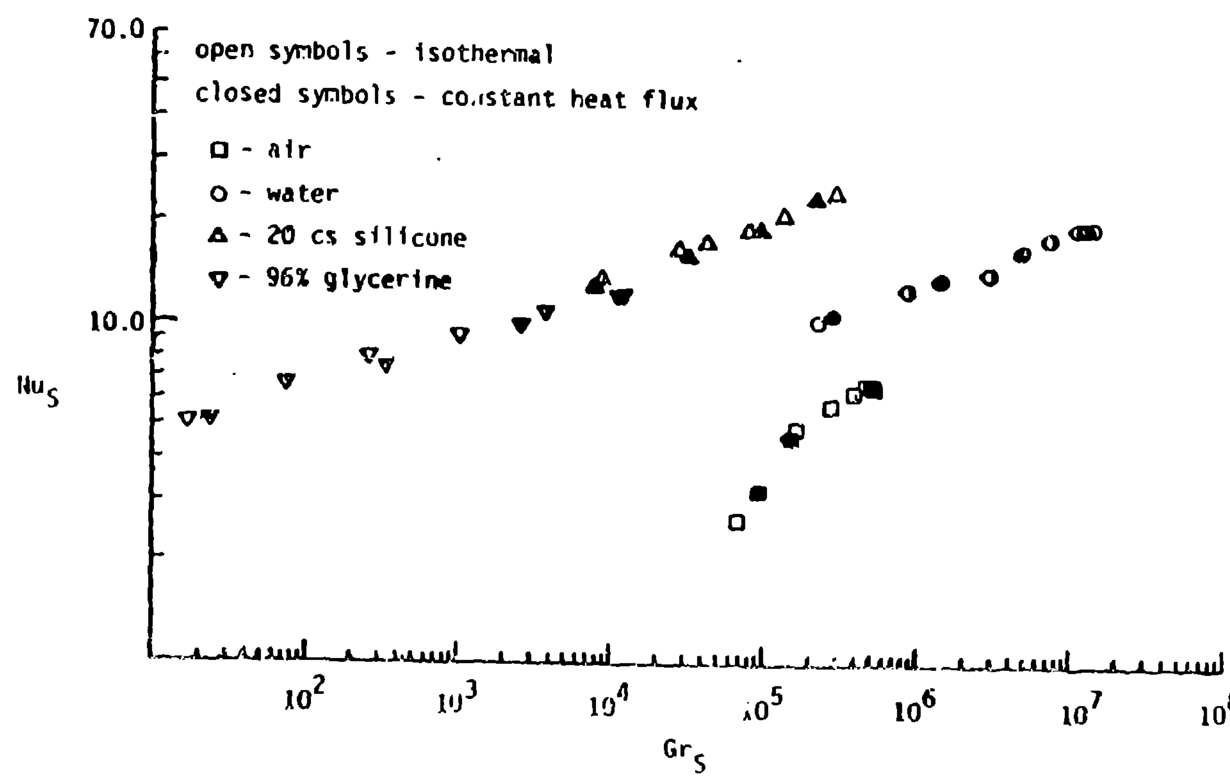

Figure 3. Comparison of the heat transfer for isothermal and constant heat flux inner body conditions using data from the eight-cylinder arrangement

RESULTS

The Rayleigh number, Nusselt number, Prandtl number, and a ratio of characteristic lengths were used in several combinations to correlate the experimental data. When calculating the Rayleigh and Nusselt numbers for use in the correlation equations, it was found that three different characteristic lengths consistently yielded the best results. The gap width, L, was defined to be the distance between hypothetical concentric spheres of volumes equal to the actual volumes of the enclosure and the cylinder arrays. R was defined to be the radius of a sphere that was equal in volume to that of the cylinder array. B was the approximate distance traveled by the boundary layer on one horizontal cylinder (assuming no flow separation). This distance was defined to be one-half of the outer circumference of a cylinder.

A comparison of isothermal to constant heat flux conditions for one cylinder arrangement is shown in Figure 3. For any one of the four fluids, the average Nusselt number of the constant heat flux data coincided very closely to the isothermal data, as was the case for the three remaining cylinder arrangements. The average Nusselt number was derived using the total amount of heat transferred by natural convection from the cylinder array to the enclosure.

Although there was no significant difference in the average heat transfer coefficient for the entire cylinder array, the local heat transfer coefficient for the cylinder rows showed a noticeable difference when changing from isothermal to constant heat conditions. When constant heat flux conditions were imposed, the upper rows of cylinders were forced to increasingly higher temperatures than that of the bottom row of cylinders. Because the enclosure caused an increase in convective activity in the upper regions of the test space, the higher temperature of the upper rows augmented the driving potential for the heat transfer, resulting in a higher local heat transfer coefficient for the upper rows.

The best correlation for all of the isothermal date was

$$Nu_B = 0.220\ Ra_B^{0.257}\ (L/R_i)^{0.660}\ Pr^{0.021}\ , \qquad (1)$$

which had an average percent deviation of 10.48. The percent deviation at a point was defined as the quantity of the absolute difference between the data value and the equation value, divided by the data value. The average percent deviation is the sum of the individual deviations divided by the number of data points. The best correlation for all of the constant heat flux data was

$$Nu_B = 0.221\ Ra_B^{0.280}\ (L/R_i)^{0.494}\ Pr^{0.016}\ , \qquad (2)$$

which had an average percent deviation of 11.03.

There were two major geometric effects evident in the experimental data. First, a staggered cylinder arrangement had a higher average heat transfer coefficient than an in-line arrangement of comparable size and spacing. Second, and more pronounced, was the increase in the average heat transfer coefficient when the spacing between cylinders increased and the total surface area of the cylinder array decreased. When the results of this study were combined with those of Crupper [5], it became apparent that an increase in the cylinder spacing led to a relative increase in the local heat transfer coefficients of the upper cylinder rows.

The geometric effect of changing the cylinder spacing and total surface area became less pronounced with an increase in the Prandtl number of the fluid medium. However, the Prandtl number of the fluid did not influence the effect of changing from an in-line array to a staggered array of comparable size and spacing. The best correlations for the air, water, 20 cs silicone, and 96% glycerine were:

$$Nu_B = 0.0097\ Ra_B^{0.207}\ (L/R_i)^{0.620}\ Pr^{-11.171}\ , \qquad (3)$$

$$Nu_B = 1.045\ Ra_B^{0.171}\ (L/R_i)^{0.712}\ Pr^{0.00084}\ , \qquad (4)$$

$$Nu_S = 0.075\ Ra_S^{0.148}\ (L/R_i)^{0.131}\ Pr^{-0.60}\ , \qquad (5)$$

and

$$Nu_S = 0.025\ Ra_S^{0.324}\ (L/R_i)^{0.360}\ Pr^{0.106}\ , \qquad (6)$$

with average percent deviations of 11.26, 5.51, 3.08, and 4.94, respectively.

All of the experimental data from this investigation are shown graphically in Figure 4, along with the data of Crupper [5] for four horizontal in-line cylinders. As shown in Figure 4, there is very little difference between the correlations for the in-line and the staggered arrangements. The best correlation for the combined data of the in-line arrangements was

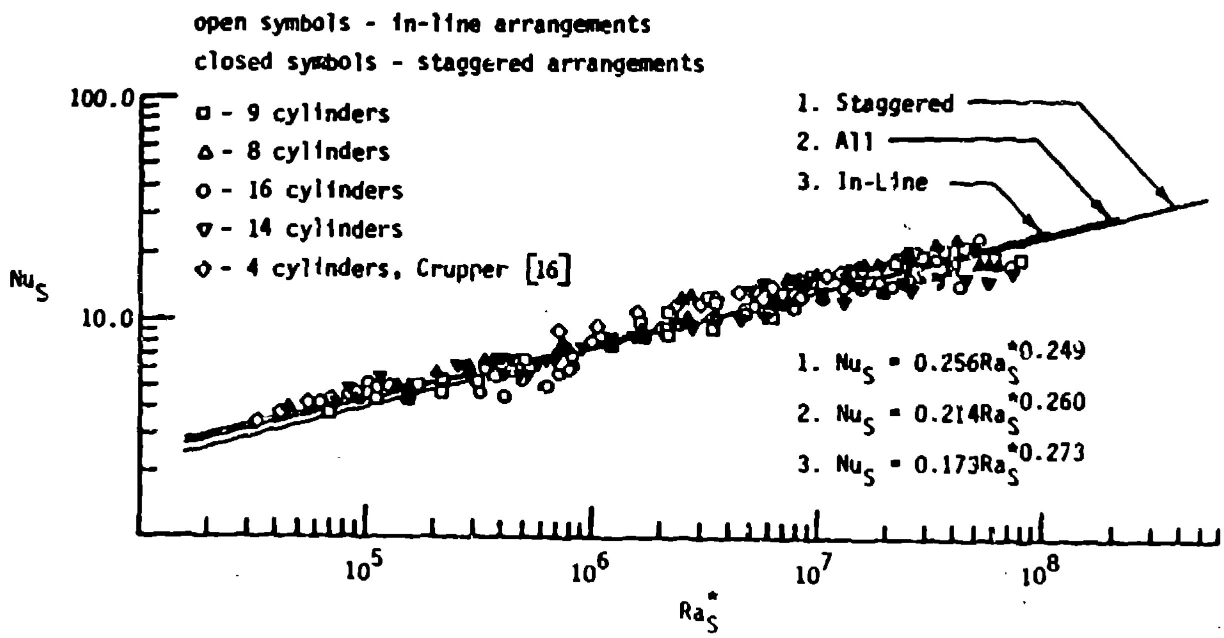

Figure 4. Heat transfer correlations for the in-line data, the staggered data, and all of the data combined

$$Nu_S = 0.174\ Ra_S^{0.209}\ (L/R_i)^{0.431}\ Pr^{0.017} \quad (7)$$

with an average percent deviation of 10.30. The best correlation for the combined data of the staggered arrangements was

$$Nu_S = 0.247\ Ra_S^{0.248}\ (L/R_i)^{0.369}\ Pr^{0.020} \quad (8)$$

with an average percent deviation of 10.70. The best correlations for all of the experimental data, based on one, two, or three correlating parameters were:

$$Nu_S = 0.2140\ Ra_S^{*0.260}, \quad (9)$$

$$Nu_S = 0.216\ Ra_S^{0.201}\ (L/R_i)^{0.366}, \quad (10)$$

and

$$Nu_S = 0.211\ Ra_S^{0.268}\ (L/R_i)^{0.351}\ Pr^{0.0189}, \quad (11)$$

where

$$0.020 \leq (L/R_i) \leq 1.041, \quad (12)$$

$$0.705 \leq Pr \leq 1.509 \times 10^4, \quad (13)$$

$$4.449 \times 10^4 \leq Ra_S \leq 1.170 \times 10^8, \quad (14)$$

$$4.633 \times 10^4 \leq Ra_S^* \leq 8.153 \times 10^7, \quad (15)$$

and they had average percent deviations of 12.00, 11.86, and 10.74, respectively.

CONCLUSIONS

This investigation has added to the amount of available data for heat transfer between multiple bodies and an enclosure. Because there was no appreciable difference in the average heat transfer coefficient for isothermal and constant heat flux inner body conditions, the applicability of the correlation results was greatly increased. However, the local heat transfer coefficients of the upper rows of cylinders, when compared to the bottom row of cylinders, were much higher for constant heat flux inner body conditions than they were for isothermal inner body conditions. The constant heat flux condition forced the upper rows to a higher temperature and augmented the driving potential for heat transfer, which resulted in higher local heat transfer coefficients for the upper rows.

The distance between cylinders and the amount of inner body surface area were the dominate factors influencing the average heat transfer coefficient. An increase in the Prandtl number of the fluid medium dampened the effects of spacing and surface area whereas the enclosure dampened the effects changing from an in-line to a staggered arrangement.

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